E. DEWEY.

The Association
of
Engineering and Shipbuilding
Draughtsmen.

POWER PUMPS

By C. EDWARDS.

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POWER PUMPS.

C. EDWARDS.

INTRODUCTION.

FOR the purposes of this paper the somewhat vague definition of Power Pumps has been extended to include similar crank-driven types of pumps, and including in that class hydraulic pumps at least as far as principles of operation are concerned.

Direct-acting steam, flywheel and borehole or deep-well types of pumps are excluded however.

The writer expresses thanks to Messrs. Ronald Trist and Company, Limited, for the use of blocks and assistance in preparing that section of the paper devoted to moulded rings.

GENERAL AND DESCRIPTIVE.

Power pumps are usually understood to be reciprocating pumps driven by belt or gear from some adjacent source of power, but if electrically and engine-driven pumps be included, the type may be considered as representative of the crank-driven reciprocating classes of force pump.

Range of Usefulness and References to other types of Pumps.

The main uses of power pumps are that of efficiently pumping :—

(*a*) Small quantities of fluid against heads varying from the lowest up to the high pressures associated with hydraulic pumps.

(*b*) Against high and low heads, but for larger quantities of fluid up to the point where centrifugal pump efficiencies are approaching the optimum possible from this class of pumps.

(*c*) Quantities usually classed as within the scope of the centrifugal pump, but against heads higher than that possible from such pumps of standard design.

(*d*) Quantities including those usually associated with centrifugal pumps, but where the variation of head at constant speed and quantity is great.

(*e*) Oils, tar, food products and viscous fluids.

(*f*) Fluids where emulsification must be guarded against.

With regard to (*a*) above, certain rotary pumps would perform this duty, but at a lower efficiency, and in this field of application all forms of positive displacement pumps are used.

Under heading (*b*), power type pumps stand out to advantage on account of high efficiency, and it is usually solely because of lower first cost that centrifugal and turbine pumps are so largely used. This is because the latter types of pumps are readily coupled direct to high-speed motors.

With reference to section (*c*), certain combinations of high lift multi-stage centrifugal or turbine pumps are able to cope successfully with heads up to 5000 feet of water. Such pumps, however, are of very special design and the resulting assembly may be more complicated than a modern high-speed type of power pump, while the efficiency of the latter would be very much greater than that of the turbine pump, which is an important consideration where large powers are in question.

Under section (*d*), it only remains to emphasise that, by virtue of their positive displacement characteristic, power type pumps automatically and instantaneously adjust to any variation in head that external conditions impose while still delivering constant capacity at constant speed. The efficiency also remains constant. Centrifugal pumps, by virtue of their peculiar characteristic, must for constant speed deliver more or less volume as the head is varied. The efficiency meanwhile varies perhaps considerably, according to the range of variation of head.

Regarding (*e*), viscous fluids, except for the lighter oils, cannot be successfully handled by centrifugal pumps owing to the excessive friction developed in passing through the internal passages of the pumps, these fluids being best handled by positive displacement pumps. Power type pumps are used wherever the highest operating efficiencies are required.

Under section (*f*), while small centrifugal and rotary pumps are used according to the viscosity of the fluid, power type pumps deal with this duty in a highly efficient manner due, of course, to the positive displacement characteristic. Recourse to low speeds is not essential, provided that the design of the valves and chambers has received full consideration.

Main Types of Pumps.—The plunger type (Fig. 1) may be considered to be the general service pump. This is because it has few working and wearing parts. The gland only may require occasional attention and this is visible and the packing easily renewable. As a single crank unit, it is very familiar as an auxiliary circulating or cooling water pump on many engines. Its use as a power pump is mainly confined to domestic and agricultural uses, on which duties it is usually belt-driven and often has suction and delivery air vessels cast with the pump barrel. Only small sizes are made, usually with built-in valve chambers. The plunger pump is also built as a two-throw pump, but it is as a three-throw pump with cranks at 120° that it is of chief importance. One of its applications is as a sewage pump, in which duty, arranged with special

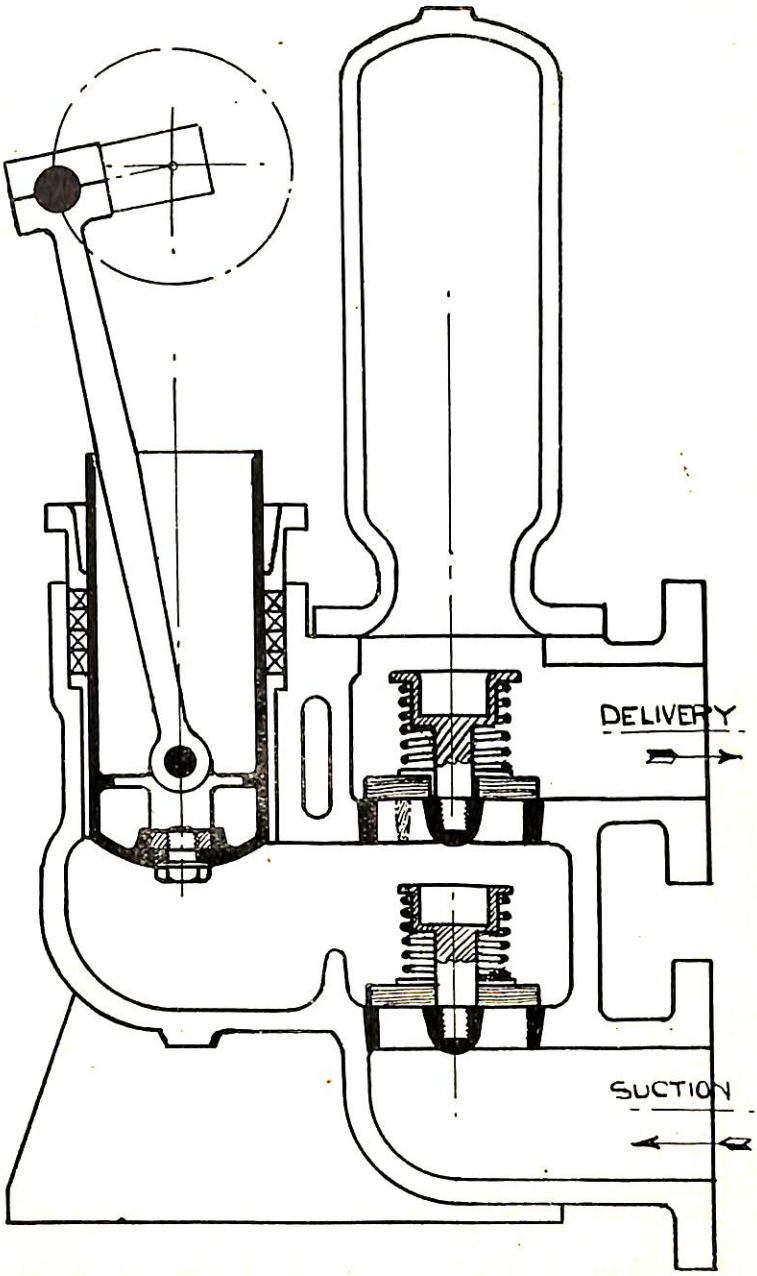


Fig. 1—Single-Acting Plunger Pump.

valves usually of the flap-type, it can deal with a large variation in head of roughly screened sewage.

An extension of the plunger towards the crank increases the range of usefulness of the pump, as it may now be fitted with an external guide, which can be lubricated, being thereby better able to take the resultant thrust due to the obliquity of the connecting rod. Fig. 2 shows this, the medium duty pump, which may be run at increased "piston" speeds compared with the ordinary plunger type.

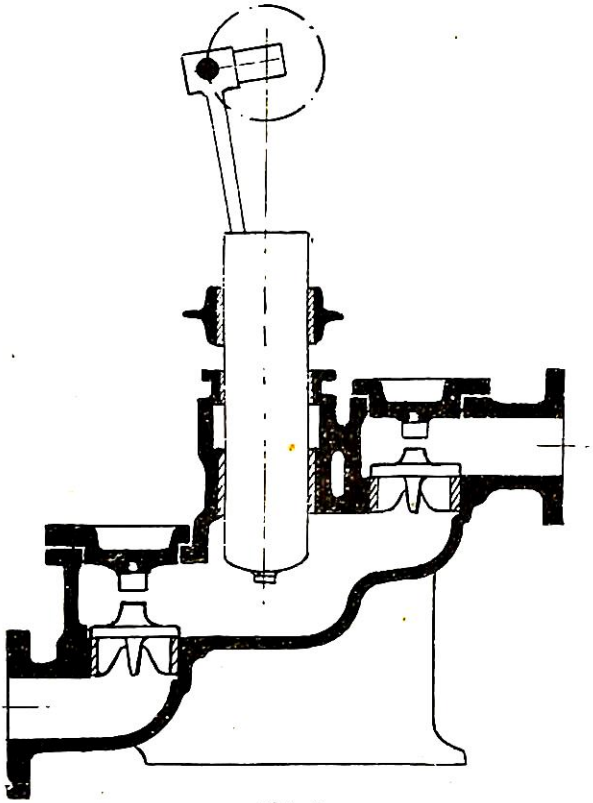


Fig. 2.

A still further improvement may be effected by providing means for taking up all the thrust of the connecting rod on to a slipper and this is shown in Fig. 3 of the ram pump.

Usually single-acting plunger pumps of one, two and three throws are arranged vertically, as thereby wear on the barrel is minimised to some extent.

Pumps of the form shown in Fig. 3 may be either horizontal or vertical. Usually they are horizontal for heavy pressures, as

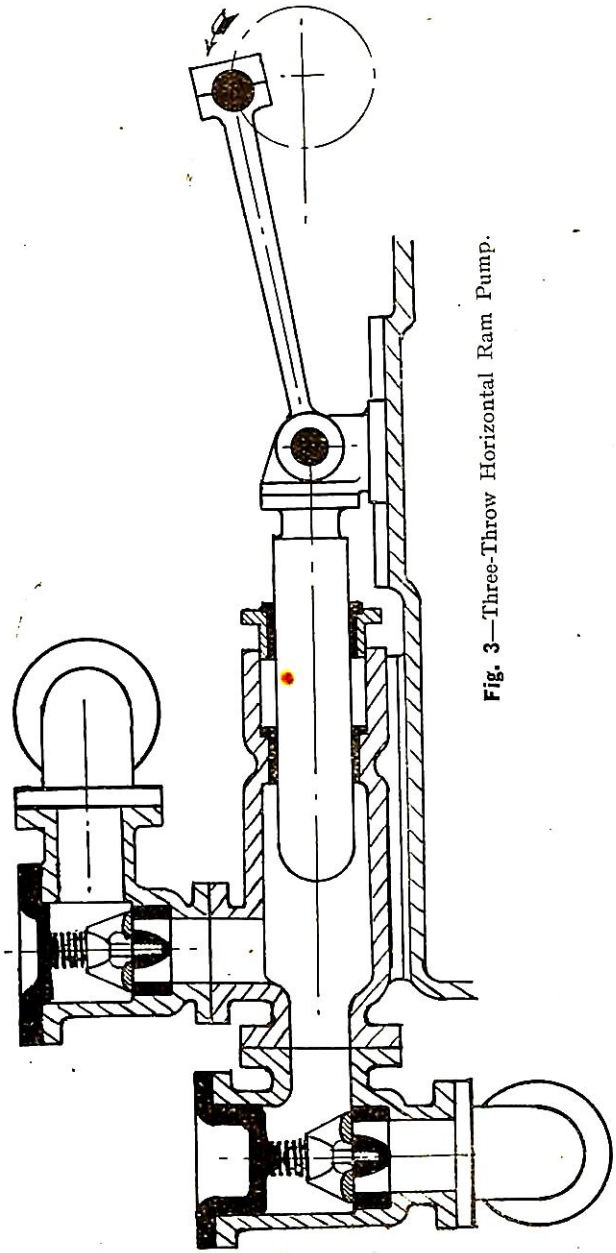


Fig. 3—Three-Throw Horizontal Ram Pump.

thereby, with the rotation as shown in Fig. 3, the forces acting on the slipper are transmitted to the ground by way of the bedplate and suction, and delivery valves are contained in chambers or pots connected to the pump barrel by short pipes.

Fig. 4 is of the double-acting bucket or piston type, which is built as a single crank machine or is duplicated to form the duplex pump with cranks at 90°. This pump has a large range of usefulness and a particular arrangement of valves will be referred to later.

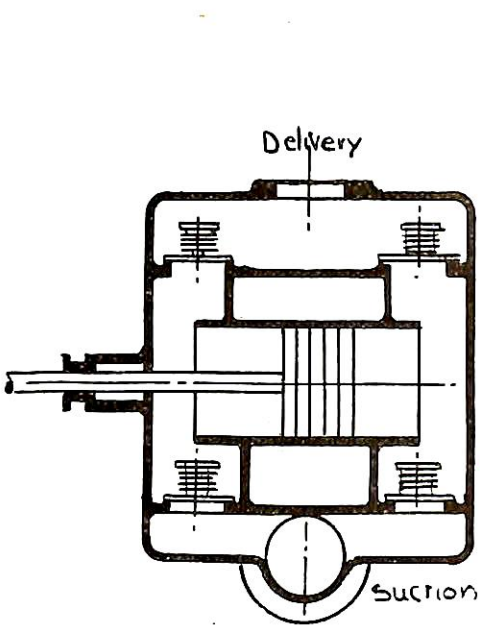


Fig. 4—Diagrammatic Arrangement of Double-Acting Pump.

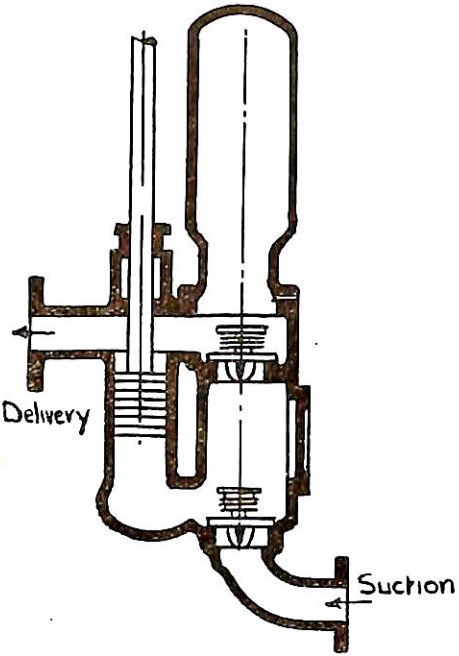


Fig. 5—Diagrammatic Arrangement of Differential Piston Pump.

The differential pump (Fig. 5). Fluid displaced by the bucket on the discharge or downward stroke passes through the delivery valve. Usually half of this volume—the exact amount being controlled by the effective area of the piston, *i.e.*, the top side in the figure—goes to occupy the space vacated by the piston, while the remaining half is delivered through the delivery pipe. On the suction stroke, however, the fluid above the piston is displaced over the delivery valve to the discharge pipe, while below the bucket a full displacement takes place through the suction valve.

There are many variations of the aforementioned types, but these are obtained often at the expense of greater complication of connecting rod, stuffing-box and valve arrangement, and while

these special pumps are suitable for particular applications, it is thought that the types mentioned are representative of the power pump types.

PUMPING HEAD.

h_{fs} and h_{fd} are the losses of head in suction and delivery pipes respectively (Fig. 9) and these may be estimated using the "Fanning" equation

$$p = \frac{\cdot 323\, f\, s\, v^2\, l}{d}$$

where p = pressure loss in lbs. ins.^{2} due to friction.
s = specific gravity referred to water at 60°F.
v = velocity in pipe ft./sec.
l = equivalent length of pipe in ft.
d = diameter of pipe inches.
f = friction factor from curve (Fig. 8).

Note.—For cold water it is usually more convenient to read the loss of head due to friction from published curves or tables; Fig. 9A will be found convenient in this respect.

For hot water, oils and viscous fluids the calculations can be made as follows :—

Obtain ν kinematic viscosity from curve Fig. 6, and specific gravity from Fig. 7, then n = absolute viscosity = νs; calculate $\frac{d\, v\, s}{n}$, estimate equivalent pipe length, equals actual length plus allowance for fittings (see Table). From Fig. 8, find friction factor f for $\frac{d\, v\, s}{n}$ and evaluate $p = \frac{\cdot 323\, f\, s\, v^2\, l}{d}$ = pressure loss lbs. ins.^{2}

or in ft. hd $hf = \frac{2 \cdot 31\, p}{s}$

Table of Allowances to be made in Pipe Fittings.

Type of Fitting.	*Equivalent Length of Pipe.*
Bends of radius 3 to 5 diameters, ...	8 diameters.
Round elbows, ...	30 ,,
Sharp elbows and tees, ...	65 ,,
Sluice valve—full open, ...	5 ,,
Globe valve, ...	150 ,,
Reflux valve, ...	35 ,,
Foot-valve and strainer, ...	55 ,,

For viscous fluids the commercial viscosity readings are in seconds, t, and these are converted into kinematic viscosity in C.G.S. units (Stokes) as follows, approximately :—

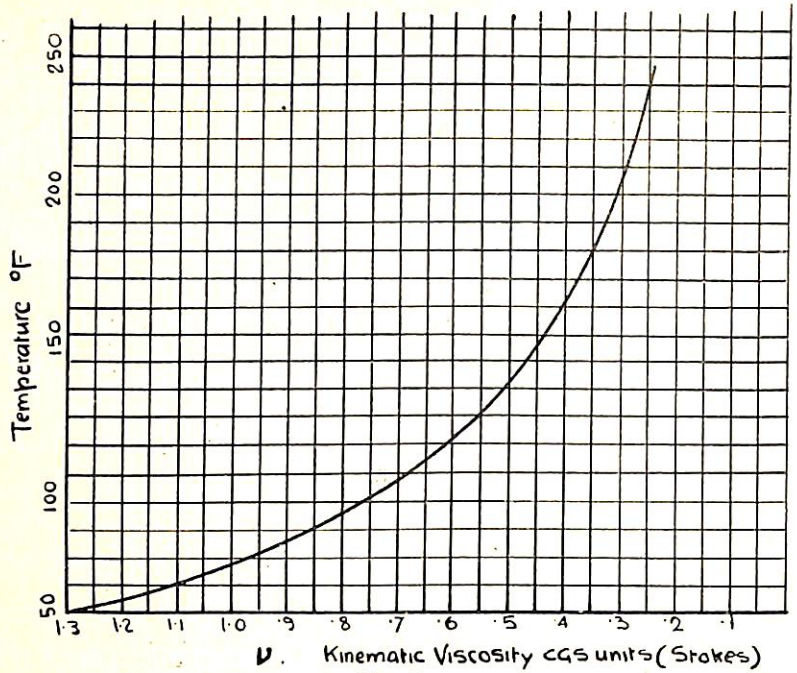

Temperature °F
250
200
150
100
50
1·3
1·2
1·1
1·0
·9
·8
·7
·6
·5
·4
·3
·2
·1
ν. Kinematic Viscosity cgs units (Stokes)

Fig. 6.

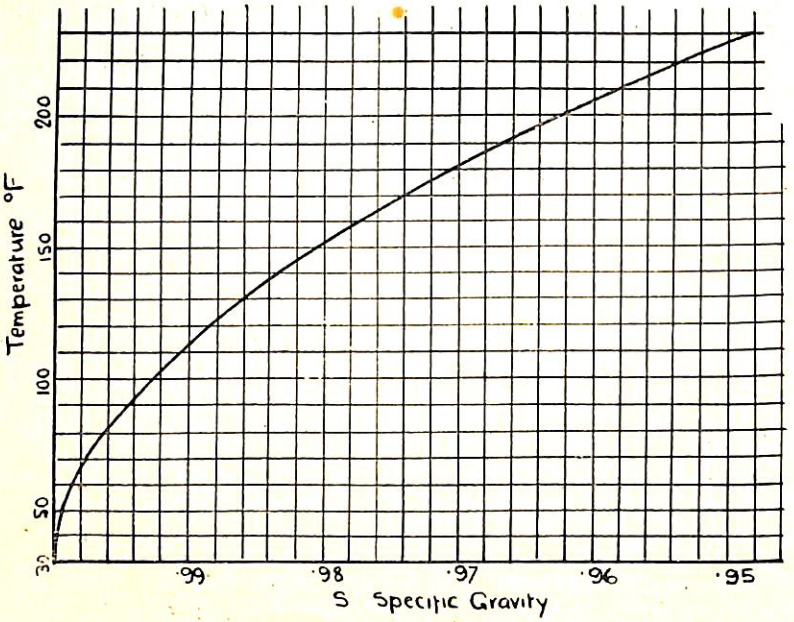

Temperature °F
200
150
100
50
·99
·98
·97
·96
·95
S Specific Gravity

Fig. 7.

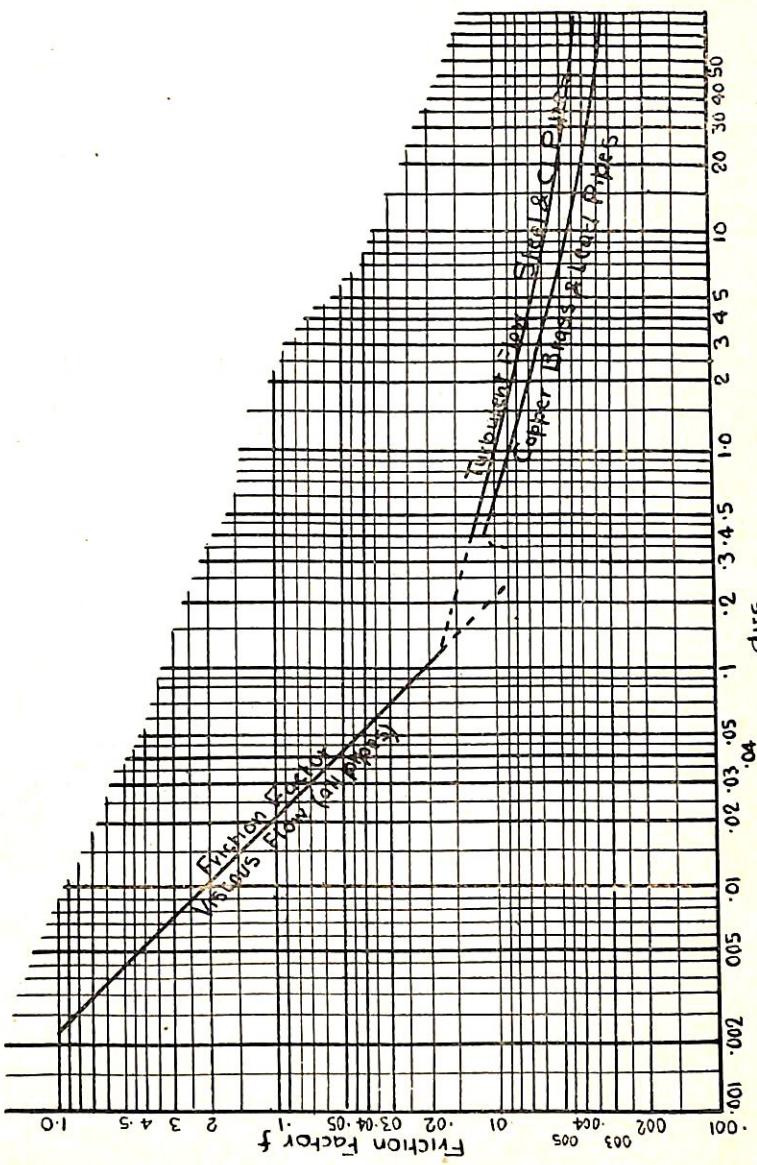

Friction Factor f
Friction Factor
Versus
Flow (all pipes)
Turbulent flow
Steel & Cast Iron Pipes
Copper Brass & Lead Pipes
dvs/n

Fig. 8.

Redwood No. 1,	t secs.	multiply by	·0026
Redwood No. 2,	t secs.	,, ,,	·0272
Engler degrees		,, ,,	·00147 × 52
Saybolt Universal,	t secs.	,, ,,	·0022

A viscosity reading refers to one particular temperature, which must be stated as well as the specific gravity. Usually a curve is supplied, showing variation of viscosity with temperature. A.E.S.D. Data Sheet, No. 73, will be found useful in this connection.

To correct specific gravity for change in temperature between 60 and 210°F., $S = S_1 - Ct$. t = temperature rise above 60° F.

where S = required specific gravity.
S_1 = known value of specific gravity at 60° F.
C = coefficient of expansion (an average value for oil = ·00037).

Note.—Fig. 9 shows the expression for total head diagrammatically and gives the components. A theoretical work diagram is also drawn, showing how the work increases for variation in delivery and/or suction head. The diagram on the right has the atmospheric lines transposed to form a single atmospheric line to coincide with the centre line of the pump, in which form it is similar to a theoretical indicator diagram.

DESIGN PRINCIPLES.

Theoretical Mean Discharge.

D = diam. of ram in ft.
L = stroke in ft.
N = number of discharge strokes per min.

The displacement or volume swept out per stroke $= \frac{\pi}{4} D^2 L$ cub. ft.

and the theoretical mean discharge

$$Q = \frac{\pi}{4} D^2 L \frac{N}{60} \text{ cub. ft. per sec.}$$

Slip.—Slip or leakage occurs past the plunger or ram, but with these parts in average condition the amount of such slip is inconsiderable. The major source of slip is the valves, which displace fluid in closing and unless synchronized with the alternate motions of the plunger, allow leakage to take place. The total slip varies from 2% to 6% in well-designed pumps for water and in this form is known as percentage slip.

Percentage Slip

$$= 100 \frac{\text{(theoretical mean discharge—actual discharge)}}{\text{theoretical mean discharge}}$$

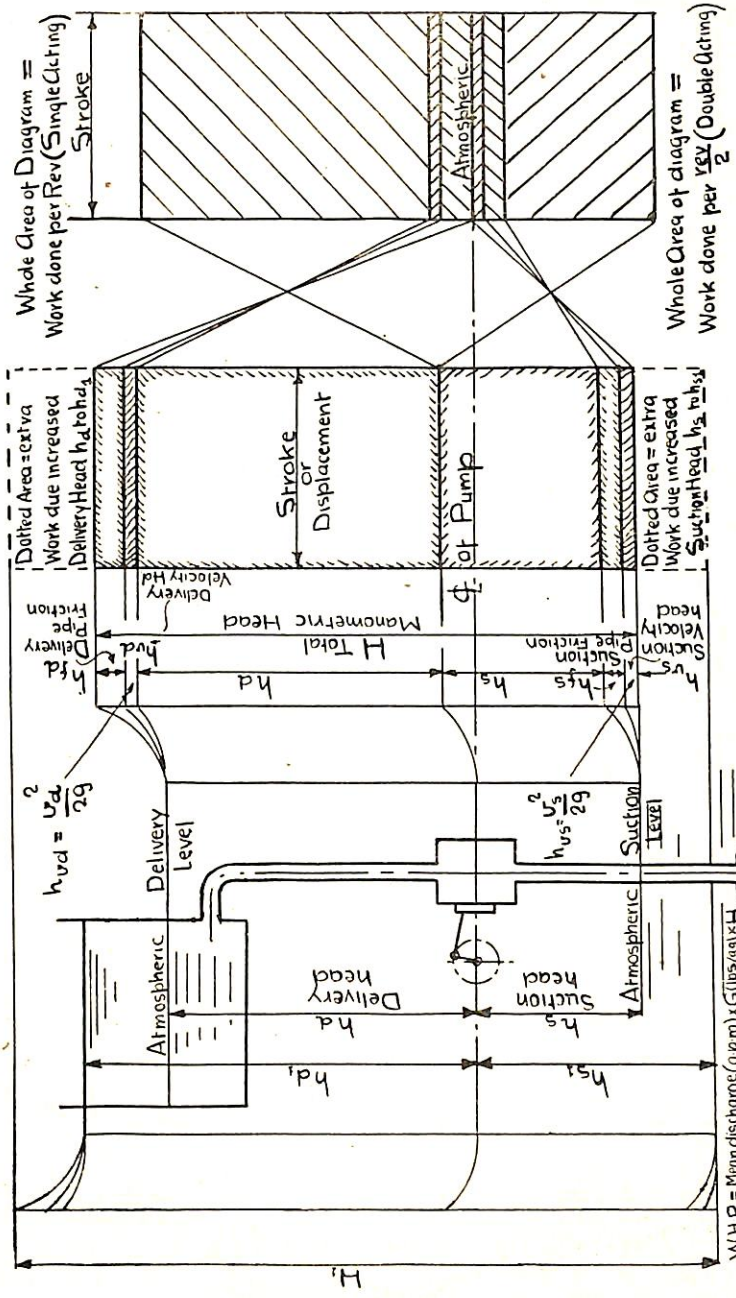


Fig. 9—Diagram of Pumping Head.

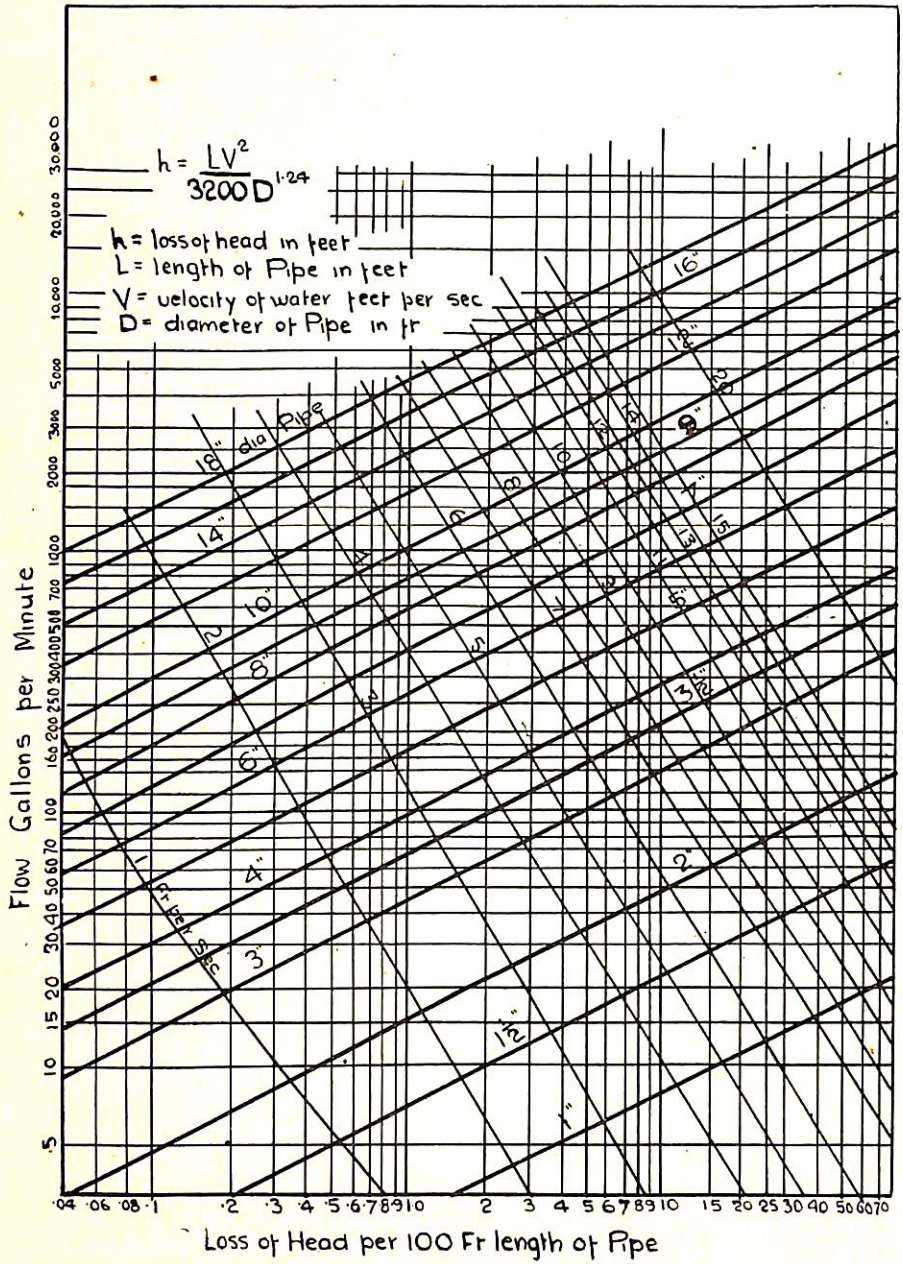

$h = \frac{LV^2}{3200D^{1.24}}$
h = loss of head in feet
L = length of Pipe in feet
V = velocity of water feet per sec
D = diameter of Pipe in ft
Flow Gallons per Minute
Loss of Head per 100 Ft length of Pipe
Pipe dia
1 Ft per Sec

Fig. 9a.

and volumetric efficiency $\% = \dfrac{\text{actual discharge}}{\text{theoretical mean discharge}} \times 100$

Mean Piston Speed.—Figures of piston speeds are often quoted, and it is pointed out that unless otherwise stated these should be taken to be

Mean piston speeds = 2 L N ft./min.

where L = stroke in ft. and N = R.P.M.

Maximum Piston Speed.—The maximum piston speed due to a crank-driven mechanism rotating at constant speed = $L \pi N$ ft./min, and this is $\pi/2$ times the mean piston speed.

Instantaneous Discharge.—The maximum instantaneous velocity in a pipe = $L \pi N \dfrac{D^2}{d^2}$ and this is $\pi/2$ times the mean velocity in the pipe which is = $2 L N \dfrac{D^2}{d^2}$

d = dia. of pipe
D = dia. of ram.

* The instantaneous velocity can be shown, as in Fig. 10, by a sine curve.

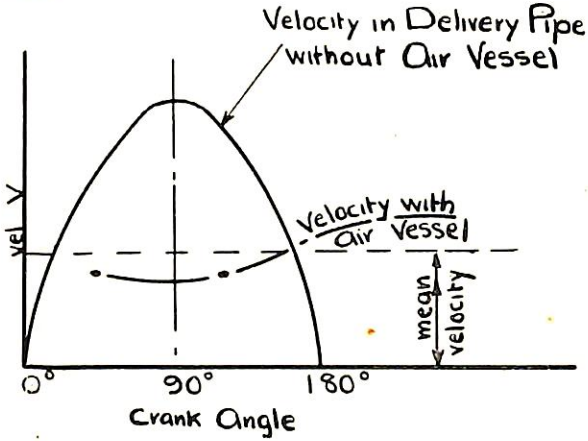


Fig. 10.

Inertia Pressure.—At the beginning and end of the stroke especially, the plunger is rapidly accelerated and retarded respectively, causing a proportionate acceleration and retardation in the suction and delivery pipes. In the case of long suction pipes the result is a sudden drop in pressure, which may cause separation

* Addison in *Practical Hydraulics.*

or cavitation of the fluid to occur when the plunger is leading or drawing the fluid. Later the fluid column may tend to overtake the retarding plunger, causing severe vibration or water hammer.

The path of the crank pin at radius R ft. is assumed to drive the plunger of the pump through the connecting rod with a simple harmonic motion, so that the velocity at any instant can be shown by a point on a sine curve. At the beginning and end of the stroke the radial acceleration of the plunger is in the line of the stroke and equals V^2/R where V is uniform linear speed of crank pin ft. per sec. and A and a represent cross sectional areas of the plunger and pipe respectively ft.^{2} units. The acceleration of the fluid in the pipe will be $\frac{V^2}{R} \frac{A}{a}$, while the mass of water in the pipe will be $\frac{w}{g} a l$,

where l = length of pipe in ft., and
w = weight of fluid in lbs. per cub. ft.
g being the acceleration due to gravity = 32·2 ft. per sec. per sec.

The accelerating force is therefore $\frac{w}{g} a l \frac{V^2}{R} \frac{A}{a}$ and this is equal to $\frac{w}{g} l \frac{V^2}{R} \frac{A}{a}$ lbs. per ft.^{2}. Calling the inertia head h_i,

$$h_i = \pm \frac{l V^2 A}{g R a} \text{ ft.}$$

the signs being used to indicate whether the plunger is being accelerated or retarded.

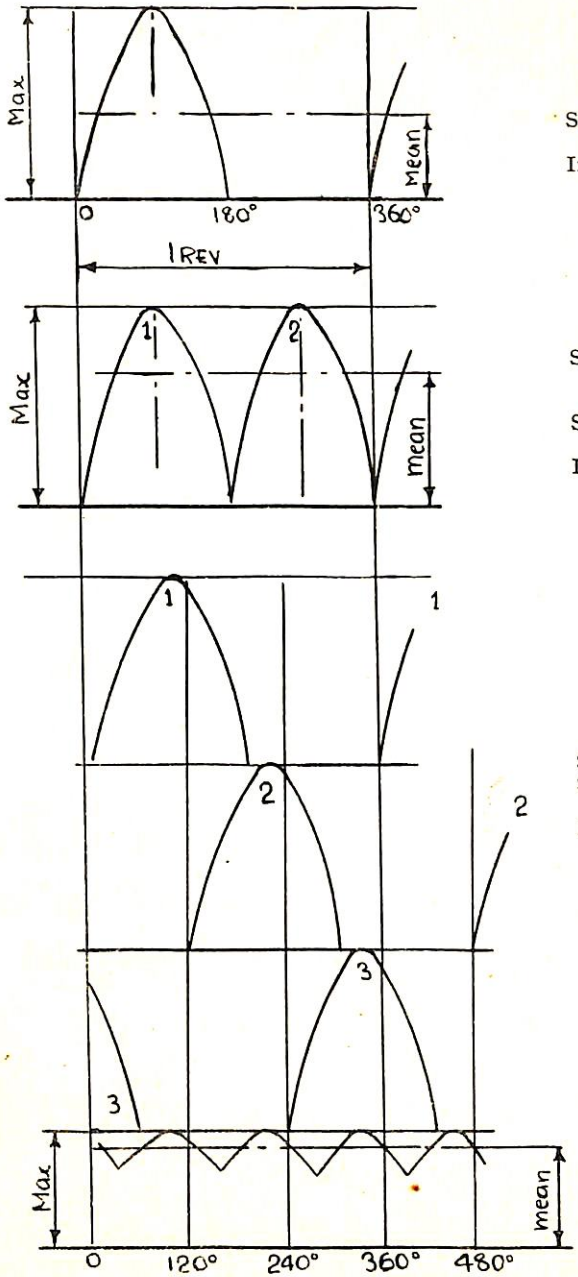
Inertia head at any instant intermediate between the limits of the stroke may also be resolved from the radial accelerated V^2/R, but this is rarely necessary, as the maximum values occur at the points of reversal of the stroke.

*Fig. 11 shows that of two pumps, one with a single large barrel and the other with smaller cylinders arranged for the same speed and discharge and without air vessels. The two-cylinder pump will generate only half the inertia pressure of the single pump, while the three-throw pump will generate one-sixth the inertia pressure of the single, due in the three-throw case to the overlapping action (see Fig. 11) of the suction or discharge.

INSTALLATION.

Fig. 12 shows an installation arrangement in sketch form. On this figure :—

*Addison in *Practical Hydraulics*.



(*a*)

Single-Acting Single-Crank.
Inertia Pressure, 100%

(*b*)

Single-Acting Twin-Crank or Double-Acting Single Crank.
Same speed and discharge as (*a*).
Inertia Pressure, 50%

(*c*)

Single-Acting 3-Throw.
Same speed and discharge as (*a*) and (*b*).
Inertia Pressure, 16.6%

Fig. 11.

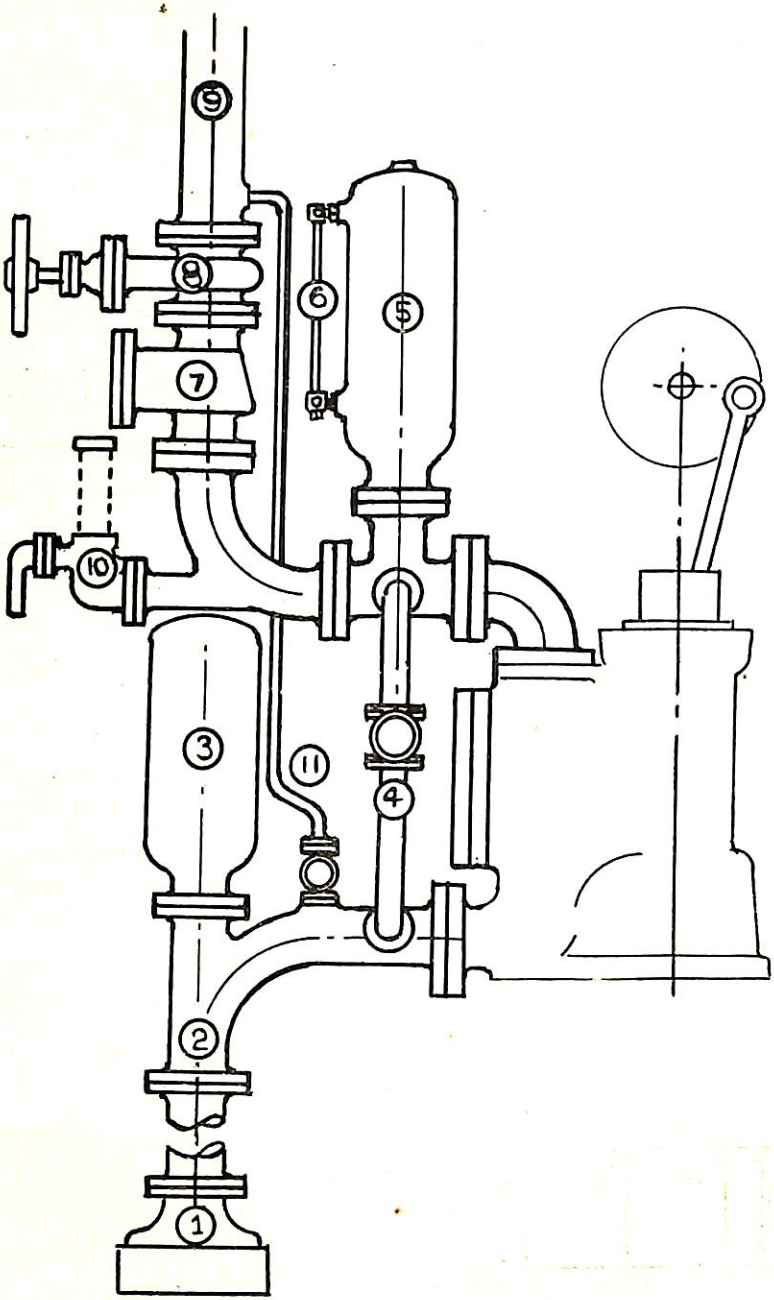


Fig. 12—Arrangement of Pumping Installation with Accessory Fittings.

(1) Is the foot-valve, often complete with strainer.
(2) Suction pipe.
(3) Suction air vessel.
(4) Delivery bye-pass and valve.
(5) Delivery air vessel.
(6) Gauge glass for air vessels.
(7) Non-return or reflux valve.
(8) Sluice valve.
(9) Delivery pipe.
(10) Relief valve.
(11) Priming bye-pass.

(1) The footvalve should be fitted wherever there is suction lift to enable the suction system to retain its water for re-starting the pump. It may not, however, be possible to fit a footvalve in certain applications, such as in pumping roughly screened sewage, where reflux action would almost certainly allow foreign matter to become wedged on the valve seat or seats, rendering these parts ineffective.

(2) Should have a gradual rise to the pump.

(3) Placed as shown, *i.e.*, directly above main suction vertical column where possible, but where long suction mains exist, should be as close to pump as possible.

(4) Used in starting to relieve the motor of the main portion of the load, otherwise excessive starting current would be used.

(5) Should be as close to pump as possible.

(6) Enables observation of level of water in air vessel to be made.

(7) Prevents reflux of the delivery into the pump and protects air vessel and delivery valves from damage.

(8) Isolates the pump from the discharge line positively when closed, and may be used to control discharge in connection with relief valve.

(10) Relief valve is a necessity on all constant delivery pumps, to protect pump from blocked discharge accidental, or if throttled, designed.

(11) Assists in connection with footvalve in rapid starting, as suction main leakage after long standing can be made good.

Additional auxiliary fittings are often called for, including :—
Air-charging arrangements. These consist of—

(*a*) Automatic snift valves or spring loaded non-return valves.
(*b*) Large pumps may have a small auxiliary air compressor for this purpose.
(*c*) Suction air or vacuum vessels on large pumps may have a small air exhauster—often of the rotary type—fitted.

Sealed valves for pumps which have to commence pumping rapidly and which work on suction lift without footvalve.

Individual installations are not fitted as a rule with all the auxiliaries mentioned, but the list includes all the alternative forms of protection and devices for promoting ease and efficiency of operation.

DESIGN DATA.

Relief Valves.—Construction is usually the concern of the valve manufacturer, and many types are available of the spring-loaded or lever-loaded types, while alarm apparatus may be incorporated where required.

Fig. 13 shows the valve and seat of a type for high pressure. This it will be noted is very similar to a common type of boiler safety valve.

The conditions of pressure and velocity may be considered in stages denoted by subscripts 1, 2, and 3—before flow commences, during flow in front of valve and at outlet from valve respectively.

Neglecting friction and shock losses,

$$H = \frac{p_1}{w} + \frac{v_1^2}{2g} = \frac{p_2}{w} + \frac{v_2^2}{2g} = \frac{p_3}{w} + \frac{v_3^2}{2g}$$

As v_1 is zero, p_1/w is the static pressure head which, acting on the valve disc, overcomes the resistance of the spring or weight, and as soon as flow is established the pressure head is reduced to p_2/w and this $= \frac{p_1}{w} - \frac{v_2^2}{2g}$. It is on this account that relief valves are often made with a surrounding lip, so as to get the maximum lifting effect from the issuing stream.

As p_3/w the outlet pressure head is zero, the stream issues with a high velocity v_3 theoretically $= \sqrt{2g \frac{p_2}{w}}$.

Frequently the lift of the valves is quite small, a matter of some thousandths of an inch, due to the high velocity.

Air Vessels.—Capacity may be derived from the following formula :—

$$*\ V = \frac{k\,A\,L}{\delta}$$ where V = mean volume of air.

Contents of air vessel in cubic feet :—

A = effective area of pump piston, bucket or plunger sq. ft.
L = length of stroke in ft.
k = a coefficient depending on the type of pump.
δ = coefficient of fluctuation equal to the ratio of the fluctuation of air pressure in the respective air vessel to the mean air pressure therein.

* B. M. Woodhouse in the *Mechanical World*.

Coefficient δ is independent of the type of pump and is equal to 1/100 for delivery air vessels and 1/10th to 1/20th for suction air vessels.

Value of k varies for different types of reciprocating pump, but is generally of equal magnitude for both suction and delivery air vessels of any one pump.

k = ·55 for single-acting, single-piston pumps.
k = ·21 for double-acting, single-piston pumps.
k = ·04 for double-acting, duplex piston pumps, with cranks at 90°,
k = ·55 for suction air vessels of differential plunger pumps.
k = ·21 for delivery air vessels of differential plunger pumps.

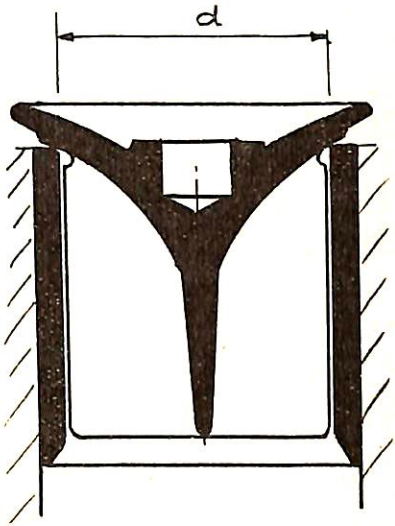


Fig. 13.

Valves.—These cannot be compared without reference to the pumping duty and conditions under which they work. Diagrams of types suited to certain duties and conditions are given in tabulated form.

Suction and delivery valves are of the well-known fluid-actuated type.

Generally speaking, it is the aim of the designer to maintain the area through the valves as large as possible, but this nearly always proves to be a matter of compromise, as large valves may prove to be sluggish in action. It can be said that the actual area of the valve seat opening below the valve will be somewhat less than the area of the barrel, while the area across the seat will usually be considerably less than this, due to limitations in the way of large lift, with consequent shock, noise and wear. Theoreti-

		(a)	(b)	(c)	(d)	(e)	(f)
Construction		Flexible Disc of Thin Metal and Guard	Flexible Rubber Disc and Guard	Thin Metal Diaphragm Spring & Stop	Thick Rubber Disc Spring & Stop	Metal Diaphragm Spring & Stop	Leather or Rubber Weighted Flap
Remarks	1	No Springs or Guides	No Springs or Guides	Centrally Guided	Centrally Guided	Centrally Guided	No Springs or Guides
	2		Low Efficiency	Efficient with Strong Spring	Efficient with Strong Spring	Good Efficiency	Poor Efficiency
	3		Quiet in Action		Quiet in Action		Wears quickly
	4	Pressure on Seat	Bending Action Wear on Seat	Pressure on Seat	Pressure on Seat	Pressure on Seat Wear on Guide	Bending of Flap
	5	Hot, Fresh, Water	Low Head Cold Salt or fresh Water	Hot Fresh Water	Cold Fresh or Salt water	Hot or Cold Clear Water	Low Head Sewage etc

		(g)	(h)	(j)	(k)	(l)
Construction		Leather or Rubber Hinged weighted Clack	Metal Wing Valve & Stop	Annular Metal Valve Spring & Stop	Rubber Composition or Metal Ball & Guide	Heavy Metal 45° Wing Valve
Remarks	1	Hinge Guide	Wing Guided	Centrally Guided	3 Arm Guide	Wing Guided, with or without Springs
	2	Fair Efficiency	Good Efficiency	High Efficiency	Fair Efficiency	High Efficiency
	3	Quiet	May be Noisy	Quiet for Small Lift	May be Noisy	Noisy with Large Lift
	4	Pressure on Seat & wear of Hinge	Wear on Guides	Pressure on Seat	Pressure on Seat	Usually in Multi-valve Assemblies as requires 40% more lift
	5	Sewage etc	Medium Head Hot or Cold clear Water	High Head Clear Hot or Cold water	Hot & Cold Oils Viscous Fluids	Hot & Cold High Head Fresh Water

Fig. 14.

cally, it is well known that the lift of a valve to maintain constant area is diameter/4, but this is rarely realised in practice. Some makers limit the lift to $\frac{1}{4}$" independently of valve size.

Considering a suction valve, at the commencement of the stroke the valve will be closed. It follows that the fluid below the valve is placed in motion, due to the action of the plunger above the valve, which, creating a negative pressure with reference to the pressure below the valve and/or to the momentum of the suction water column, causes the valve to rise off its seat. The fluid flows through the valve, which is a passage of varying cross-section (unless lift $= d/4$), with the outlet area less than the inlet, and there may be a change of head due to friction, shock, and flow through the varying cross-section, so that the change of head produces a pressure difference across the valve resulting in a force being available for lifting the valve—once flow is established—and this can be utilised to a greater or less degree to control the closing of the valve.

Inlet velocity Fig. 15 = v, and outlet velocity v_3.

v_1 = velocity over seat, all ft. per sec.
p = inlet pressure, lbs. per ft.^{2}.
p_1 = outlet pressure, lbs. ft.^{2}.
d = mean diameter of valve ft.
W = weight of valve.

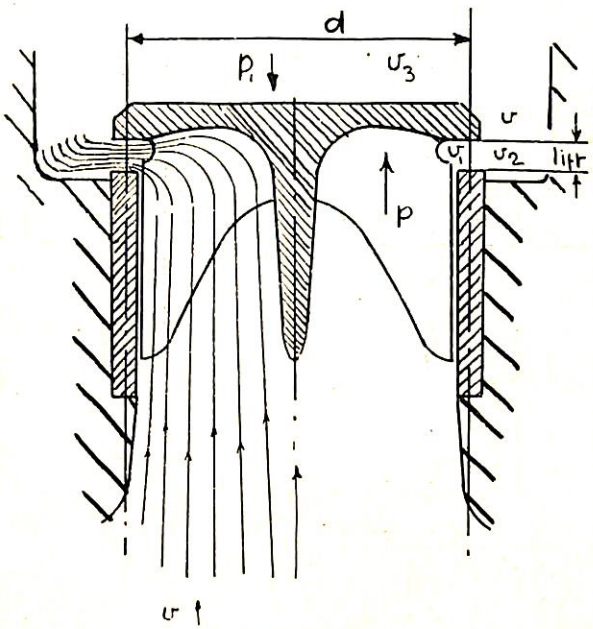


Fig. 15.

Velocity ft. per sec.		1						2						3						4					
Diam. of Valve ins.		1	2	3	4	5	6	1	2	3	4	5	6	1	2	3	4	5	6	1	2	3	4	5	6
Lift of Valve in terms of Diameter.	$d/8$	·02	·08	·18	·32	·51	·73	·08	·32	·73	1·3	2·0	2·9	·18	·73	1·6	2·9	4·6	6·6	·33	1·3	2·9	5·2	8·2	11·8
	$d/16$	·139	·55	1·2	2·2	3·4	5	·55	2·2	4·9	8·8	14·8	20	1·2	5	11·4	20	31	45	2·2	8·8	20	33	55	…
	$d/32$	·68	2·7	6·1	11	17	24	2·7	11	24	43	…	…	6·1	25	…	…	…	…	11	…	…	…	…	…
	$d/64$	2·9	11·7	26	47	74	106	12	47	…	…	…	…	26	…	…	…	…	…	…	…	…	…	…	…
	$d/8$	·51	2·0	4·5	8·1	13	18	·73	2·9	6·6	12	18	26	1	4	8·9	16	25	36	1·3	…	…	…	…	…
	$d/16$	3·4	15	30	55	86	125	4·9	20	44	80	…	…	6·6	27	60	…	…	…	8·9	…	…	…	…	…
	$d/32$	17	…	…	…	…	…	24	…	…	…	…	…	34	…	…	…	…	…	…	…	…	…	…	…
	$d/64$	…	…	…	…	…	…	105	…	…	…	…	…	…	…	…	…	…	…	…	…	…	…	…	…
Velocity ft. per sec.		5						6						7						8					

Fig. 16.—Effective Closing Force for Various Diameters of Valve at Different Velocities.

Total gain of head in passing through the valve is calculated to be $-v^2 - 4{\cdot}9\ v_1^2 + 6{\cdot}25\ v_1\ v$, from which W has been calculated for various openings, sizes and velocities (Figs. 16).

W is the upward force which balances the weight of the valve.

Note—$v_3 = \frac{1}{2}\ v$.

W is also the effective closing force, and it will be seen that this becomes progressively smaller as the velocity through the valve is reduced. As the velocity through the valve is proportionate to the plunger velocity, it is evident that a heavy valve may commence to close, owing to the decreasing plunger velocity, before the end of the stroke, and this device is often resorted to in connection with limited valve lift and multiple valves to obtain increased speed in operation and efficiency.

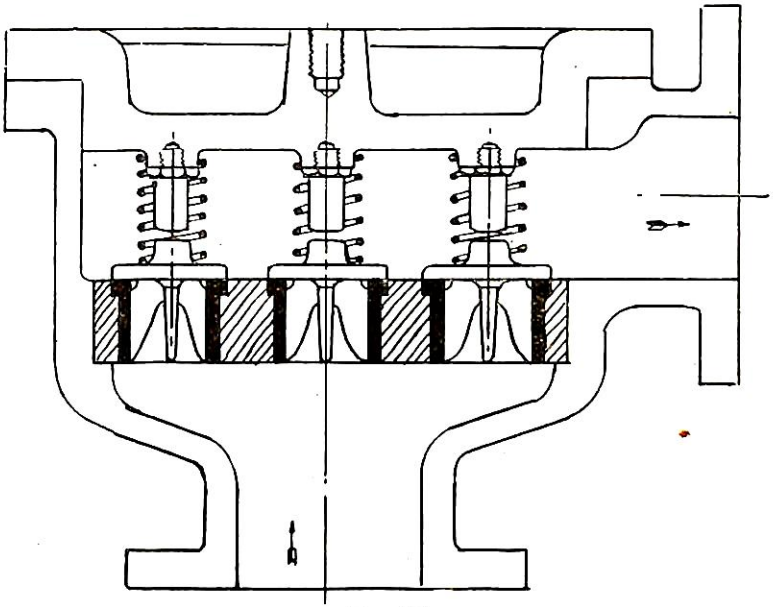


Fig. 17.

Unwin and Mellanby, in *Elements of Machine Design*, quote a relation attributed to Bach, which shows that for any given valve the limit of speed at which valve closes quietly is given by an equation of the form n^2S = constant, where n = strokes per minute, and S = stroke, and at that limit the effective closing force = $n^2S \times$ constant.

Attempts to evolve a mechanically-operated valve have been made, notably by Professor Riedler, and although very small reciprocating pumps are sometimes used having a form of mechanically-operated valve—usually a port covered and uncovered by the plunger—the fluid-actuated valve continues to be used.

With regard to the tabulated types of valves (Fig. 14), they have been drawn up to show compliance with certain requirements which are given as :—

(1) Simplicity. Valve must function for long periods out of sight and without attention, and lubrication will be only that which can be obtained from the working fluid, so that relative sliding and swivelling motions are to be deprecated.
(2) Efficiency, with which is included slip and accurate synchronization of opening and closing with motion of plunger. Tightness when closed.
(3) Wear in service and noise.
(4) Limit.
(5) Suitability for particular liquids in service.

Multiple Valves.—Fig. 13 shows an arrangement of multiple valves in a unit chamber. The valves shown are of the wing type, but almost any form of valve may be adopted for use in the same manner.

SUCTION LIFT, SEALING STUFFING BOXES AND PISTONS.

Suction Lift.—All types of power pumps will deal with considerable suction lift when fitted with a footvalve. Fig. 18 gives the maximum suction lift at various temperatures and altitudes, and is reproduced from Kent's *Mechanical Engineers' Handbook*. The figures to the right of the thick line show the suction head required for dealing with hot water. This table represents good practice for single-acting three-throw pumps and double-acting single and duplex pumps. Single-acting, single-barrel pumps have suction lift capacities rather less than the above, usually 70-85% of the values given in the table.

Self-Charging Pumps.—Most standard types of power pump will not start pumping without being first primed. For this duty the clearance volume of the pump should be the absolute minimum, certainly less than the plunger displacement. Valves should be tight and if possible of rubber, and sealing spaces should be the minimum possible. These conditions are best served by the double-acting piston or bucket type of pump, either single or duplex. This type of pump is arranged with the valves as shown in Fig. 19, instead of below the pump barrel, which is the cheapest form of construction. By this means the barrel will be full of water on starting, thus sealing and lubricating the piston, stuffing-box and valves in their duty of exhausting air from the suction system.

Theoretical Lift at any Altitude = barometer in inches $\times$ 1·132 for water, and for other liquids, divide by specific gravity of liquid.

Altitude above sea level in feet.	Temperature °F.															
	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210
	Suction Lift										Suction Head.					
0	22	20	17	13	13	11	8	6	4	2	0	3	5	7	10	12
2,000	19	17	15	13	11	8	6	4	2	1	3	5	7	10	12	15
4,000	17	15	13	10	8	6	4	1	1	3	5	7	10	12	14	...
6,000	15	13	11	8	6	4	2	1	3	5	7	10	12	14	16	...
8,000	13	11	9	6	4	2	0	3	5	7	9	12	14	16	...	...
10,000	11	9	7	4	2	0	2	4	7	9	11	14	16	18	...	...

Fig. 18.

Actual lift may be taken as ·6 to ·82 of theoretical, according to class of pump.

Stuffing Boxes.—Plunger pumps are almost universally packed with a number of turns of hemp, flax or Egyptian cotton soaked in tallow, or a special graphited packing of self-lubricating nature may be preferred. There is also usually a neck-ring on to which the packing is pressed down by the gland, which may be bushed.

Ram pumps, too, are sometimes sealed as above, but under modern conditions it is usually found that a moulded ring packing of special section, the action of which is automatic, is preferable. It should not be overlooked that these special sealing rings depend on pressure for their successful working, and that on this account,

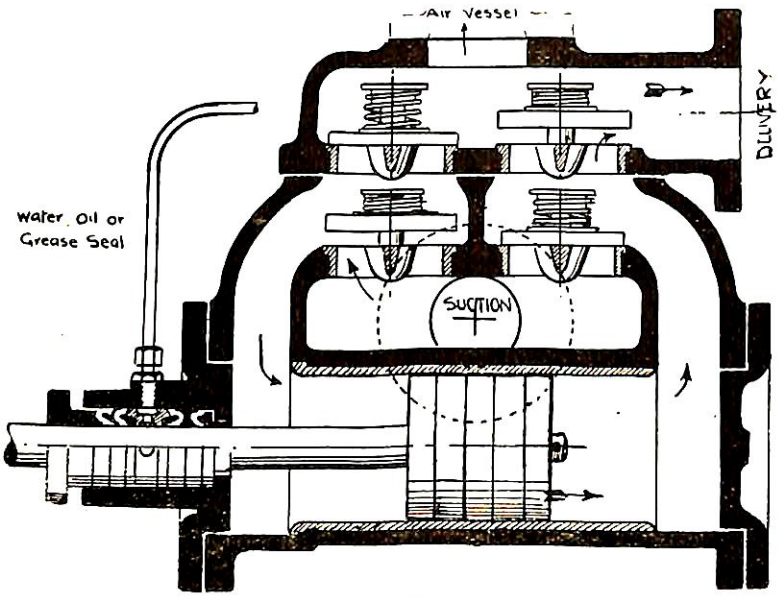


Fig. 19.

in some cases as, for instance, where a single ring is employed, it may be an advantage to have grooves or channels machined to lead the fluid behind the flexible tongue of the ring. Figures 20 and 21 show moulded ring gland packings fitted to stuffing boxes of ram pumps, Fig. 20 being the normal arrangement for pressures up to 500 lbs. ins.^{2}, while Fig. 21 shows metal supports as used on high pressures.

The familiar Bramah or "U" leather was formerly used for this application and is still largely employed.

Double-acting pumps may have packing as previously described for plunger pumps or moulded rings may be used. Fig. 19 shows

the stuffing box of a double-acting pump arranged for sealing under pressure due to high suction lift conditions.

Buckets and pistons of double-acting pumps are sealed with, or by means of piston rings of metal or special moulded composition rings, in which the fluid is introduced by way of special ducts to the underside of the ring, which is thereby expanded into sealing contact with the barrel. Cotton packing was also largely used until recently for many applications, but the present-day tendency is to use formed rings of metal, leather or synthetic material bonded with fabric.

Fig. 23 shows the piston of a vertical double-acting power pump, and is the design of Messrs. Ronald Trist and Company, Limited.

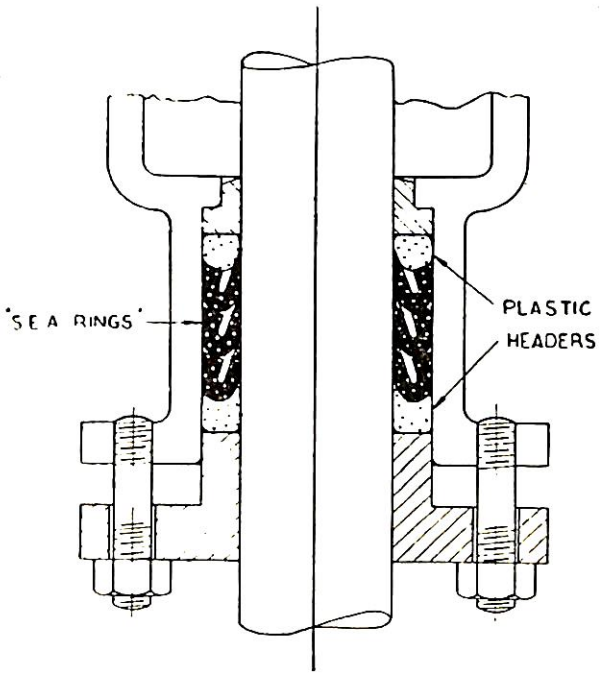


Fig. 20—Standard arrangement with plastic headers. Gland and neck bush may be flat or bevelled.

PROPORTIONS OF IMPORTANT PARTS.

P = thrust on plunger, bucket or ram = ·7854 D^2p lbs., where D = plunger diameter and p = pressure in barrel lbs. per ins.^{2}. Length of connecting rod = five times crank throw. Maximum thrust due to obliquity of connecting rod = ·196 P.

The following formulae are taken from *Pumps, Construction and Maintenance*, by Owen A. Price, M.I.M.E., in Pitman's *Engineering Educator* :—

Plungers.—Compressive stress, 8000 and 4500 lbs. ins.^{2} for cast-iron and gunmetal respectively.

Piston rods of double-acting pumps :—Tensile strength, 8000 and 6000 lbs. ins.^{2} for mild steel and bronze respectively, for smallest section usually at the nut.

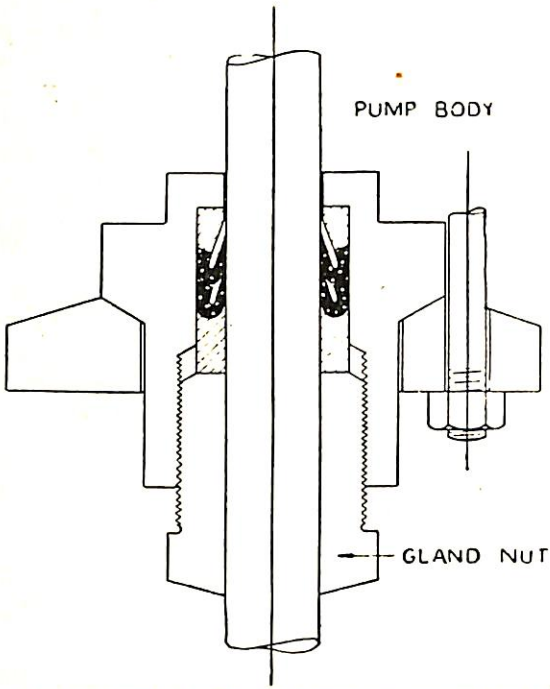


Fig. 21 —With a threaded gland nut a separate header should be used. Note the thread is on a larger diameter than the stuffing box.

Diameter of body of rod, calculated as for strut or using $d_1 = \cdot 025\ D\sqrt{p}$, where d_1 = rod diam. inches, D = piston diam. inches, p = maximum pressure difference between the two sides of the piston, lbs. ins.^{2} Neglecting inertia bending forces, the mid-diameter of connecting rod $d_2 = \cdot 037\ \sqrt{P^{\frac{1}{2}} \times L}$ ins., where L = length of connecting rod ins.

Crankshaft.—Diameter $= C\ \sqrt[3]{\frac{SHP}{N}}$ where SHP = maximum shaft H.P., N = R.P.M. and C = constant, having values from 5 to $7\frac{1}{2}$ for light and heavy pressure pumps respectively.

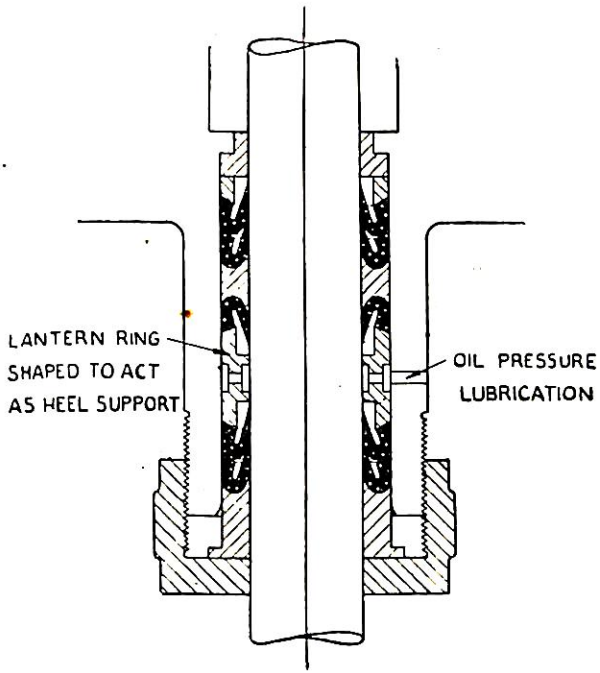


Fig. 22—How to pack a gland with pressure seal.

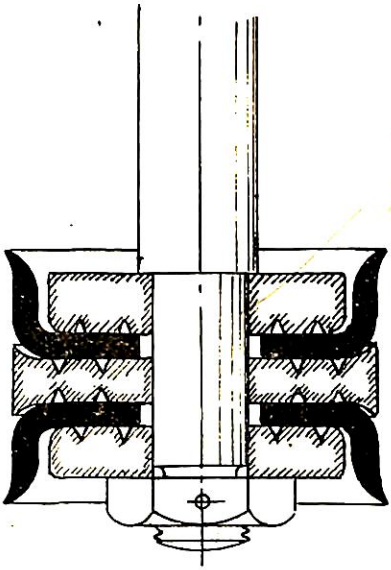


Fig. 23.

Bearing Pressures.

Crank pin, 600 to 750 lbs. ins.^{2}
Crosshead pin, 1000 to 1250 lbs. ins.^{2}
Main bearing, 400 to 600 lbs. ins.^{2}
Crosshead slipper, 40 to 50 lbs. ins.^{2}

The above are calculated from the maximum load and the projected bearing area.

Circular Forms of Castings.—Thickness may be calculated, using Barlow's empirical rule applicable to both thick and thin cylinders :

$$2tf = p\ (D + 2t)$$

where f = safe working tensile stress lbs. ins.^{2}
t = thickness of wall, ins.
D = internal diameter, ins.
p = internal pressure, lbs. ins.^{2}
f = 1500 to 2000 lbs. ins.^{2} for cast iron.
f = 4000 to 5000 lbs. ins.^{2} for cast steel, or, alternately,
f = 2800 lbs. ins.^{2} for cast iron ;
f = 5000 lbs. ins.^{2} for cast steel ;
but increase wall thickness for two latter by $\frac{1}{4}''$ and $\frac{1}{8}''$ for cast iron and cast steel respectively.

Water Velocities.—Normal mean velocities representing common practice :—

Suction pipe,	3 to 3·5	ft./sec.
Suction valves,	2·5 to 4·5	ft./sec.
Delivery pipe,	3·5 to 4	ft./sec.
Delivery valves,	3 to 5	ft./sec.
Plunger chamber,	1·5 to 2	ft./sec.

Around ram and valves, 5 ft./sec.

Loads on Valves.—Load on valve disc $P = \frac{\pi}{4} d^2 p$, where d = mean diameter, Fig. 15, and p = pressure difference across the valve.

Area of seat = $\pi d w$, where w = projected width of seat, and $w = \frac{d p}{4 f_c}$, where f_c = crushing stress of the material.

Unwin and Mellanby, in *Elements of Machine Design*, give the following values for f_c :—

C.I., 1000 lbs. ins.^{2},
Phosphor bronze, 3000 lbs. ins.^{2},
Gun metal, 2000 lbs. ins.^{2},

and the same authority gives for rubber valves :—

Bearing pressure on rubber in contact grid face not greater than 40 lbs. ins.^{2}.

Width of seat of flap or clack valves should be not less than $\frac{1}{5}\sqrt{D} + \frac{1}{8}''$.

45° wing valves and metal valves, projected bearing width, may be $\frac{1}{10}\sqrt{D} + \frac{1}{16}$" minimum, or may be calculated from the load as previously given.

Shaft H.P.—The sources of loss are volumetric, *i.e.* slip; hydraulic, *i.e.* shock and friction losses in valves and pump barrel and friction in connecting or bridge pipes; mechanical, *i.e.* friction between packing and rod, plunger or ram, friction in crank and connecting rod assembly.

$$\text{Shaft horse power, SHP} = \frac{\text{WHP}}{\text{Pump efficiency}}$$

Pump efficiencies may be as high as 90%.

$$\text{Brake horse power, BHP} = \frac{\text{SHP}}{\text{Gear efficiency}} = \frac{\text{WHP}}{\text{Overall effi'cy}}$$

MISCELLANEOUS DESIGN NOTES.

Notes *re* Lubrication, Stuffing Box Friction, Springs, Slip, Motors, Byepass Testing, Materials, Design.

Drip-feed and oil-bottle lubricators are commonly used, but grease lubrication may be used to advantage where ball and/or roller bearings are used for rotating and oscillating bearings.

Forced-feed lubrication is being largely adopted on the largest electrically-driven pumps.

Many pumps run without external lubrication to the stuffing box, relying entirely on the lubricating qualities of the packing, and this may be necessary for clean water or food products. "U" leather packings for clean liquids and high pressures may be lubricated by neatsfoot, sperm, etc., oils.

Goodman's formulae for frictional resistance of "U" leathers is :

$F = \cdot 08\ P + C$, where F = frictional resistance lbs. ins.^{2} of water pressure ;

P = water pressure, lbs. ins.^{2} ;

$$C = \frac{100}{D} \text{ to } \frac{250}{D}$$

D = inches diameter of ram.

Machinery's *Encyclopaedia* gives "spring pressure for discharge valves = ·005 to ·01 water pressure, with a maximum pressure of 5 lbs., and for suction valves ·25 to ·5 lbs. Slip may be proportional to total valve area and $\sqrt{\text{lift}}$."

Considerable power may be required to accelerate power pumps from rest, due to journal and stuffing box friction. On this account alone, squirrel cage motors will be prohibited and wound rotor motors are the rule for this class of pump. Further, owing to the

constant delivery characteristic, starting against normal pumping head will entail a very high starting torque, and on this account a delivery byepass valve is used which allows the delivery to be short-circuited to suction and the load applied more gradually by closing the byepass valve.

Pumps for sewage, oil and viscous fluids are tested with water, then at constant speed weight of fluid delivered per min. = weight of water delivered per min. × specific gravity of fluid and R.P.M. for fluid

$$= \frac{\text{R.P.M. for water}}{\text{Specific gravity of fluid at operating temperature}}$$

for a constant speed electric motor, A.C. or D.C., the input or electrical H.P. $= \frac{\text{Volts} \times \text{amps}}{746}$ and BHP = electrical H.P. × efficiency of motor. KW. consumption in B.O.T. units $= \frac{\text{BHP} \times \cdot 746}{\text{Efficiency}}$

Current consumption D.C. motors, amps. $= \frac{\text{BHP} \times \cdot 746}{\text{Volts} \times \text{efficiency}}$

Amps per phase for 3-phase motors

$$= \frac{\text{BHP} \times 746}{\text{Volts} \times \text{efficiency} \times \text{power factor} \times 1 \cdot 732}$$

Fig. 24 gives full load speed of A.C. motors, also average efficiencies and power factors.

Materials.—Where materials in contact are immersed in sea water or other corrosive fluid, choice should be made by reference to the electro-chemical series, choosing materials as close together in the series as possible. Intensity of corrosion is high in case of small area of steel in contact with large area bronze, conversely, corrosion not so rapid for large area of steel contacting with a small area of bronze. Stainless steel in contact with ordinary steel will accelerate attack on the mild steel in sea water and alloy steel is badly affected in contact with bronze in the same medium. Monel metal and bronze together are highly resistant to sea water.

Practice in Pump Construction.—All parts C.I. for ammoniacal liquor, beet juice, lubricating and fuel oil, tar, naptha, paint, creosote, drainage. All parts G.M. for sea water (Admiralty practice), weak acids, vinegar, sugar, syrups.

C.I., G.M. lined, for waterworks, mines, sea water (commercial practice), beer wort, sewage, petroleum, tanneries, domestic.

In special cases, water velocities in the pump may be increased considerably above those values previously given. Velocities may

Full Load Speeds of A.C. Motors.

No. of Poles.	Cycles per Second.								
	25	27	30	33⅓	40	42	50	60	100
2	1440	1560	1730	1920	2300	2420	2880	...	...
4	720	780	865	960	1150	1210	1440	1725	2880
6	480	520	575	640	770	807	960	1150	1920
8	360	390	433	480	575	605	720	862	1440
10	290	310	345	385	460	485	575	690	1150
12	240	260	288	370	385	403	480	575	960
14	205	222	247	275	330	346	412	492	822
16	180	195	216	240	287	302	360	431	720
18	160	171	192	214	256	268	320	384	640
20	142	156	173	192	230	242	287	345	575

Full Load Speed, R.P.M. $= \left(\frac{\text{Cycles} \times 120}{\text{Poles}}\right) - 5\%$ Slip.

Efficiency and Power Factor of Various Sizes A.C. Motors.

B.H.P.	Efficiency %			Power Factor.		
	Load.			Load.		
	1	¾	½	1	¾	½
1	76	73	68	·8	·76	·7
2-5	81	79	75	·84	·81	·75
5-10	84	82	78	·87	·84	·79
10-25	86	85	82	·89	·86	·82
25-40	88	87	84	·90	·88	·84
50-75	89	88	86	·91	·89	·85
75-100	91	90	88	·92	·9	·87

Fig. 24.

increase as $\sqrt{H}$, thus if velocity was 3 ft. per sec. for 100 ft. head at 400 ft. velocity would be $\frac{\sqrt{400}}{\sqrt{100}} \times 3 = 6$ ft. per sec., while the efficiency remains constant. Velocity through the valves may be increased to twice this figure in special instances, but pipe velocities must be kept low, except for the very shortest pipes.

In design mean piston speed will be the first aim, and consistent with the limits of inertia pressure previously mentioned, will be as high as possible up to 500 ft. per min. in special cases, in order to keep the pump as small as possible, as in fire pumps.

The relation of this class of pump to other types will be watched in selecting a pump for a particular duty.

The position as regards the centrifugal and/or turbine pump has been stated at the commencement of the paper. It should be borne in mind in this connection that the power pump is suited to continuous duty over long periods without overhaul and with scarcely any alteration in efficiency. The efficient operation of many rotary pumps depends very largely on fine working clearances however.

It should not be overlooked, too, that small glandless reciprocating pumps can be built to compare favourably with the smallest pumps as regards speed of rotation and size, and it is probably in this direction that major developments will occur in the future.

WORKED EXAMPLE.

A pump is required for 100 gallons per minute of clean cold water against a total head of 500 ft. from a well, lowest water level 20 ft. below ground, which is approximately at sea level.

It is required to fix the size of the pump, the speed of rotation, and H.P. to drive.

Note.—As suction lift high, pump will need to be close to well with shortest possible suction pipe ; a suction air vessel should not be necessary provided the suction pipe can be kept short, say 22 ft. long. Further, for 500 ft. head, a ram pump with its continuously circular cross-section of pump barrel would be required. In this connection it should be noted that double-acting piston pumps are of a generally weaker construction, due to large areas which are of a more or less flat shape.

Assume a low velocity in suction pipe of 3 ft. per sec., then diameter of suction pipe

$$= \sqrt{\frac{100}{60} \times \frac{1}{6{\cdot}23} \times \frac{1}{3} \times \frac{144}{{\cdot}7854}} = 4''$$

From Fig. 18, maximum suction lift at 60°F. = 22 ft. It will be necessary to check over the suction conditions to see that this figure is not greatly exceeded.

Assume 22 ft. of straight pipe - - -	22 ft.
,, 1 bend from page 9 = 8 × 4/12 -	$2\frac{2}{3}$,,
,, 1 footvalve and strainer from page 9 = 55 × 4/12 - - - -	$18\frac{1}{3}$,,
Equivalent length of pipe, - - - -	43 ft.

and from chart, Fig. 9A, friction = 1·1/100 × 43 = ·473 ft.

Velocity head = $3^2/64{\cdot}4$ = ·14 ft., and assuming barrel 18″ above ground, total suction lift = 20 + 1·5 + ·473 + ·14 = 22·1 ft., and this is sufficiently close to the maximum of Fig. 18 to be allowed.

When the pump is at rest, the absolute pressure head in the barrel = 12·5 ft., and assuming that at commencement of suction stroke, separation due to inertia would take place at zero absolute head, then 12·5 = inertia head.

$$\text{Inertia head} = H_i = \frac{l\,V^2\,A}{g\,R\,a}$$

$$\text{but } V = 2\,\pi\,\frac{RN}{60} \text{ and } \frac{V^2}{R} = \frac{RN^2}{92},$$

also, if the stroke be kept equal to 1·5 × diameter of ram,

$$D = 1{\cdot}33\,R \text{ and } D^2 = 1{\cdot}77\,R^2$$

$$\text{and } H_i = \frac{l\,RN^2 \times {\cdot}7854 \times 1{\cdot}77\,R^2}{32{\cdot}2 \times 92 \times \frac{{\cdot}7854}{144}\,d^2}$$

$$\text{from which } R^3 = \frac{H_i\,d^2}{{\cdot}086\,l\,N^2} \qquad (1)$$

Also the theoretical mean discharge per barrel = Q cubic ft./sec. and

$$Q = {\cdot}7854\,D^2 \times 2\,\frac{RN}{60}$$

and substituting 1·77 R^2 for D^2 as before, then

$$Q = \frac{{\cdot}7854}{30} \times 1{\cdot}77\,R^3N \qquad \text{and } R^3 = \frac{Q}{{\cdot}046\,N} \qquad (2)$$

Solving (1) and (2),

$$N = \frac{H_i\,d^2}{1{\cdot}87\,Q\,l}$$

where N = R.P.M.
H_i = inertia head.
d = diameter of suction pipe in ins.
and l = length of suction pipe in ft.

then for a single barrel,

$$Q = \frac{100}{60} \times \frac{1}{6{\cdot}23} = {\cdot}268 \text{ cusecs.}$$

$$\text{and } N = \frac{12{\cdot}5 \times 16}{1{\cdot}87 \times {\cdot}268 \times 22} = 18{\cdot}3 \text{ R.P.M.}$$

while for the same discharge and three throws, Q per barrel $= {\cdot}268/3 = {\cdot}089$ cusecs and N = 55 R.P.M.; and substituting in (2),

R = ·329 = 3·95″ and stroke = 7·90″, diam. = 5·27″ = 7·9″/1·5.

As the above are rather odd sizes, 5″ bore × 7·5″ stroke may be tried for capacity, then

Theoretical mean discharge

$$= {\cdot}7854\ D^2L \frac{N}{60} \times 3 \qquad D = {\cdot}416 \qquad L = {\cdot}625$$

Also 100 gals. per min. will be increased to 103 gals./min., allowing for 3% slip.

$$\text{then } \frac{103}{60 \times 6{\cdot}23} = {\cdot}7854 \times {\cdot}416^2 \times {\cdot}625 \times \frac{N}{20} \text{ and N} = 65 \text{ R.P.M.}$$

Checking over inertia head for these conditions,

$$12{\cdot}5 = \frac{22\ V^2 \times 25}{32{\cdot}2 \times {\cdot}312 \times 16} \text{ from which V} = 1{\cdot}91 \text{ ft. per sec.}$$

$$\text{and R.P.M.} = \frac{1{\cdot}91 \times 60}{3{\cdot}142 \times {\cdot}625} = 58{\cdot}5$$

and this shows that a 5″ × 7½″ pump should not be run at a greater speed than 58·5 R.P.M., whereas 65 R.P.M. is required for this example. Trying 6″ × 9″ for inertia,

$$12{\cdot}5 = \frac{22\ V^2 \times 36}{32{\cdot}2 \times {\cdot}375 \times 16} \text{ and V} = 1{\cdot}75$$

therefore, R.P.M. = 44·6; and checking over from the capacity,

$$N = \frac{103 \times 20}{60 \times 6{\cdot}23 \times {\cdot}7854 \times {\cdot}25 \times {\cdot}75} = 37{\cdot}6 \text{ R.P.M.}$$

37·6 R.P.M. is, therefore, the required speed, the size of the pump being 6″ bore × 9″ stroke.

Note.—Re-bore and stroke. The figures given, viz. :—stroke = 1·5 × bore, are common in power pump practice, as they represent a compromise between the short stroke, high-speed pumps having equal bore and stroke with increased slip and the long-stroke pumps of bore equals half stroke, having a higher volumetric efficiency and a slower speed of rotation.

$$\text{Water H.P.} = \text{W.H.P.} = \frac{100 \times 10 \times 500}{33{,}000} = 15{\cdot}2$$

and with volumetric efficiency of 97% and pump efficiency of 87%,

$$\text{SHP} = \frac{15{\cdot}2}{{\cdot}97 \times {\cdot}87} = 18,$$

and with 97% gear efficiency, the B.H.P. absorbed at the pump coupling

$$= \frac{18}{{\cdot}97} = 18{\cdot}5$$

The overall efficiency is therefore $\frac{15{\cdot}2}{18{\cdot}5} = 82\%$

S.H.P. = Shaft horse power.

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